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Shapiro et al.

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(54) **TWO-STAGE LINEAR COMPRESSOR**

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H02P 1/00 (2006.01)

(52) **U.S. Cl.** **62/175**; 62/228.3; 62/510; 62/513; 417/44.1; 417/417; 318/135

(58) **Field of Classification Search** 62/175, 62/510, 228.3, 113, 513; 417/415, 417, 44.1, 417/410.1; 318/135
See application file for complete search history.

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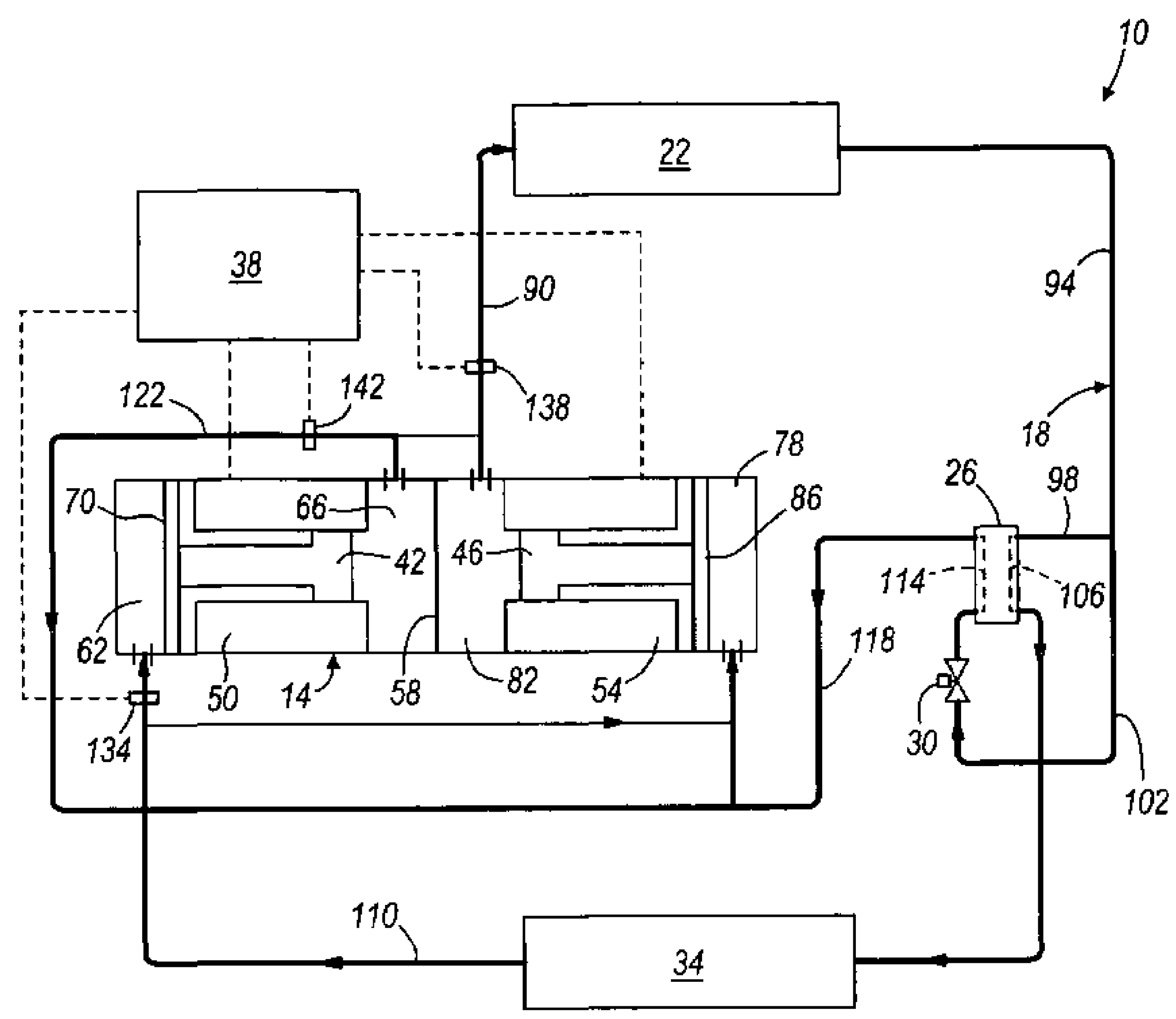
(57) **ABSTRACT**

A refrigeration system includes a two-stage linear compressor having a first piston disposed in a first cylinder and a second piston disposed in a second cylinder. The linear compressor is operable in an economizer cycle wherein the first piston operates as a first stage of the economizer cycle and the second piston operates as a second stage of the economizer cycle. A controller is coupled to the linear compressor to control a volume flow ratio of the linear compressor. The controller stores a plurality of coefficients of performance for a range of particular operating conditions of the linear compressor and each coefficient of performance corresponds to a desired volume flow ratio and a desired secondary evaporating temperature. Based upon measured operating conditions of the linear compressor, the controller determines a highest coefficient of performance from the plurality of coefficients of performance and varies operation of at least one of the first and second pistons to achieve the desired volume flow ratio.

29 Claims, 4 Drawing Sheets



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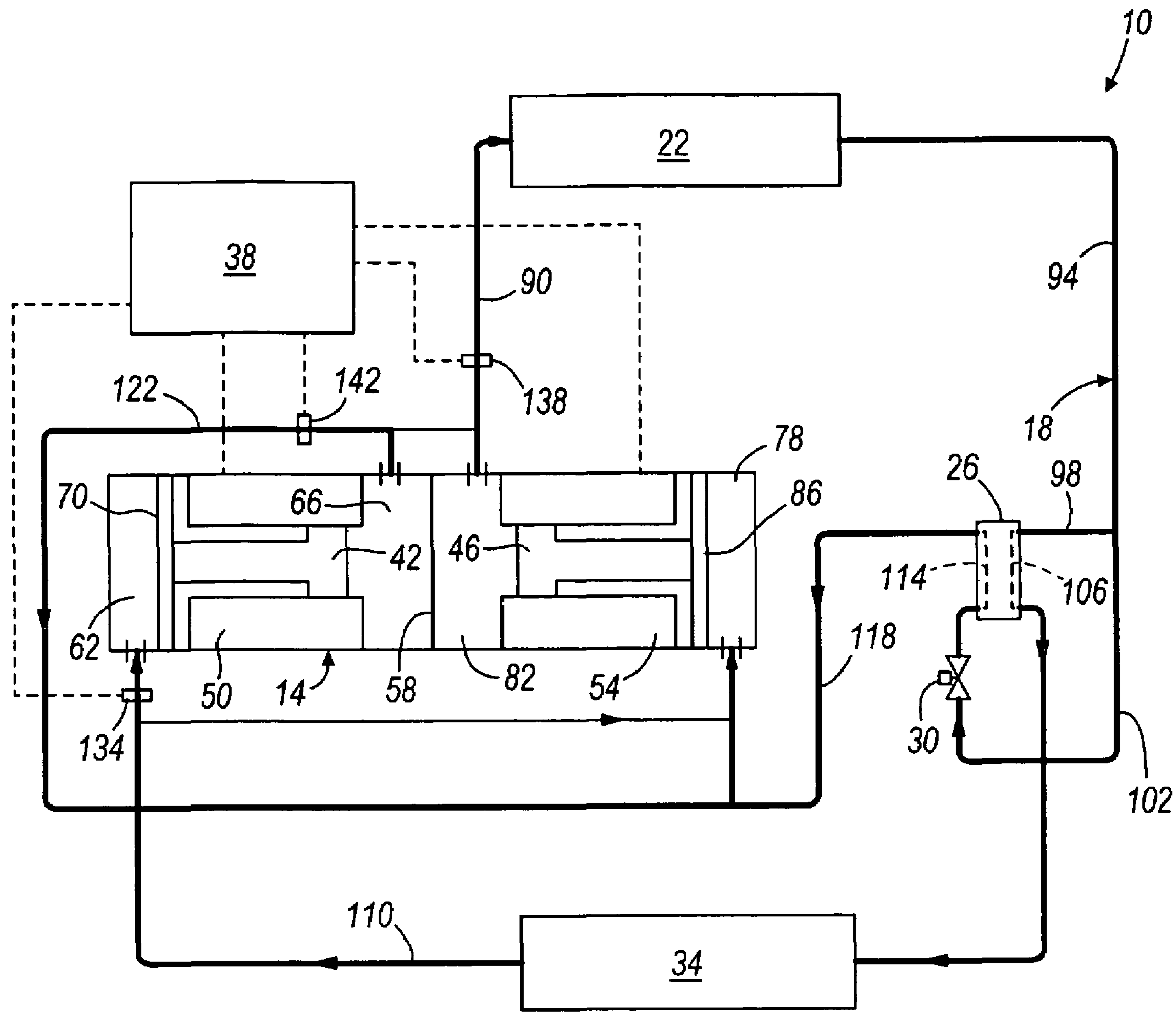
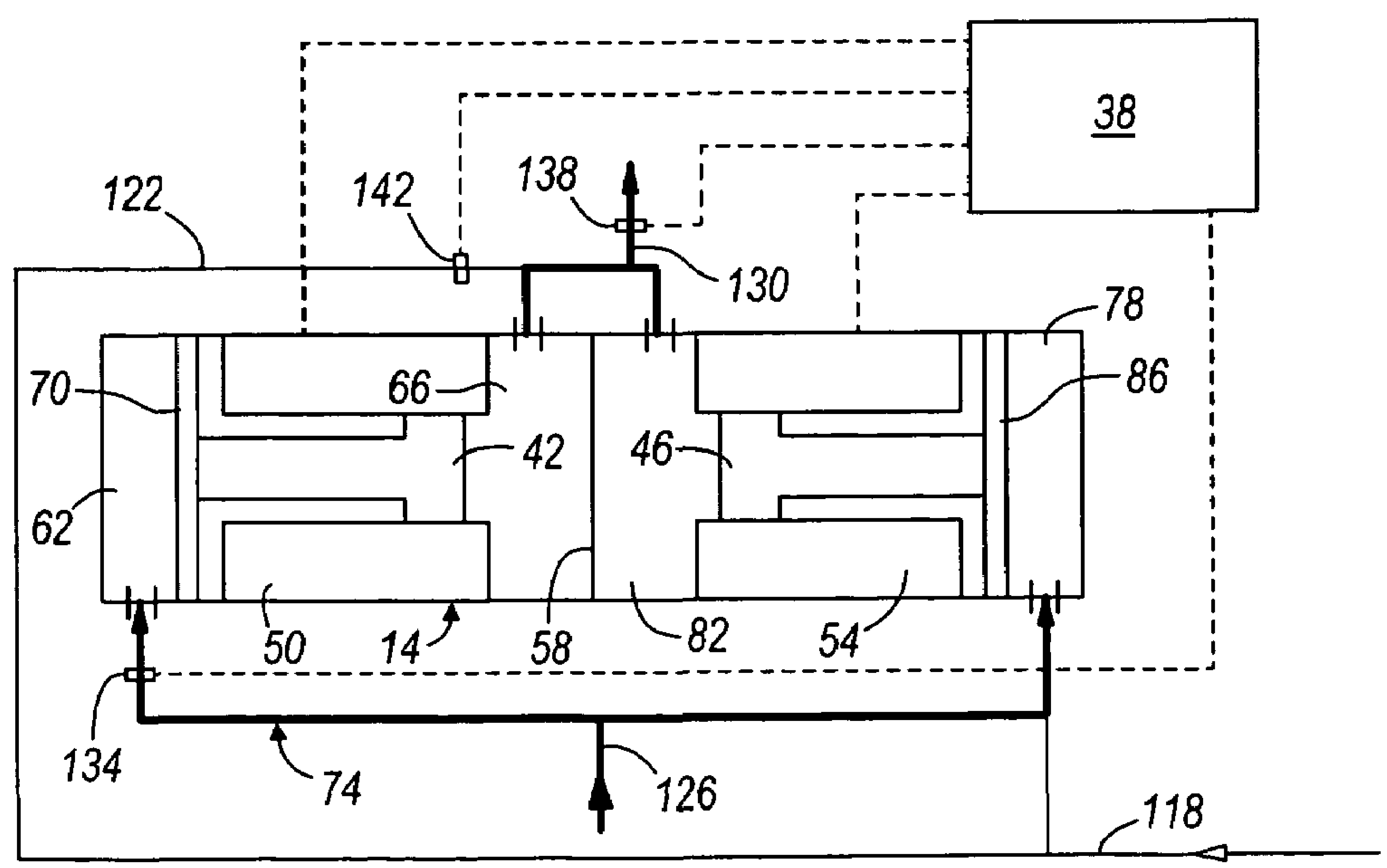

10
22
38
90
94
122
142
138
18
78
26
98
70
66
42
46
86
114
106
62
118
30
134
50
14
58
82
54
102
110
34

FIG. 1


38
122
142
138
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78
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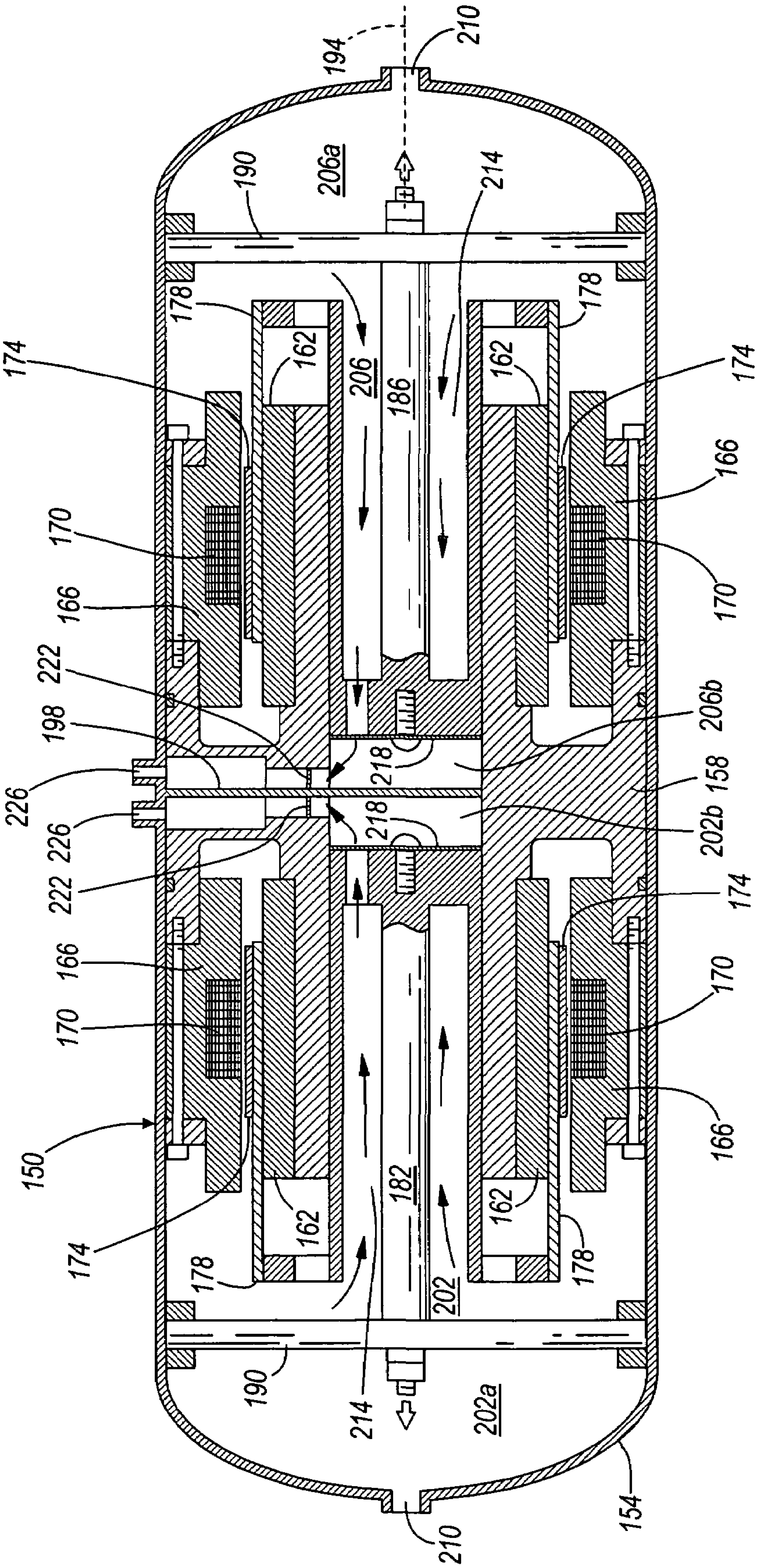
66
86
42
46
62
50
14
58
82
54
134
74
126
118

FIG. 2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190
214
178
174
162
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222
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206b
202b
158
150
182
202
202a
154

FIG. 3

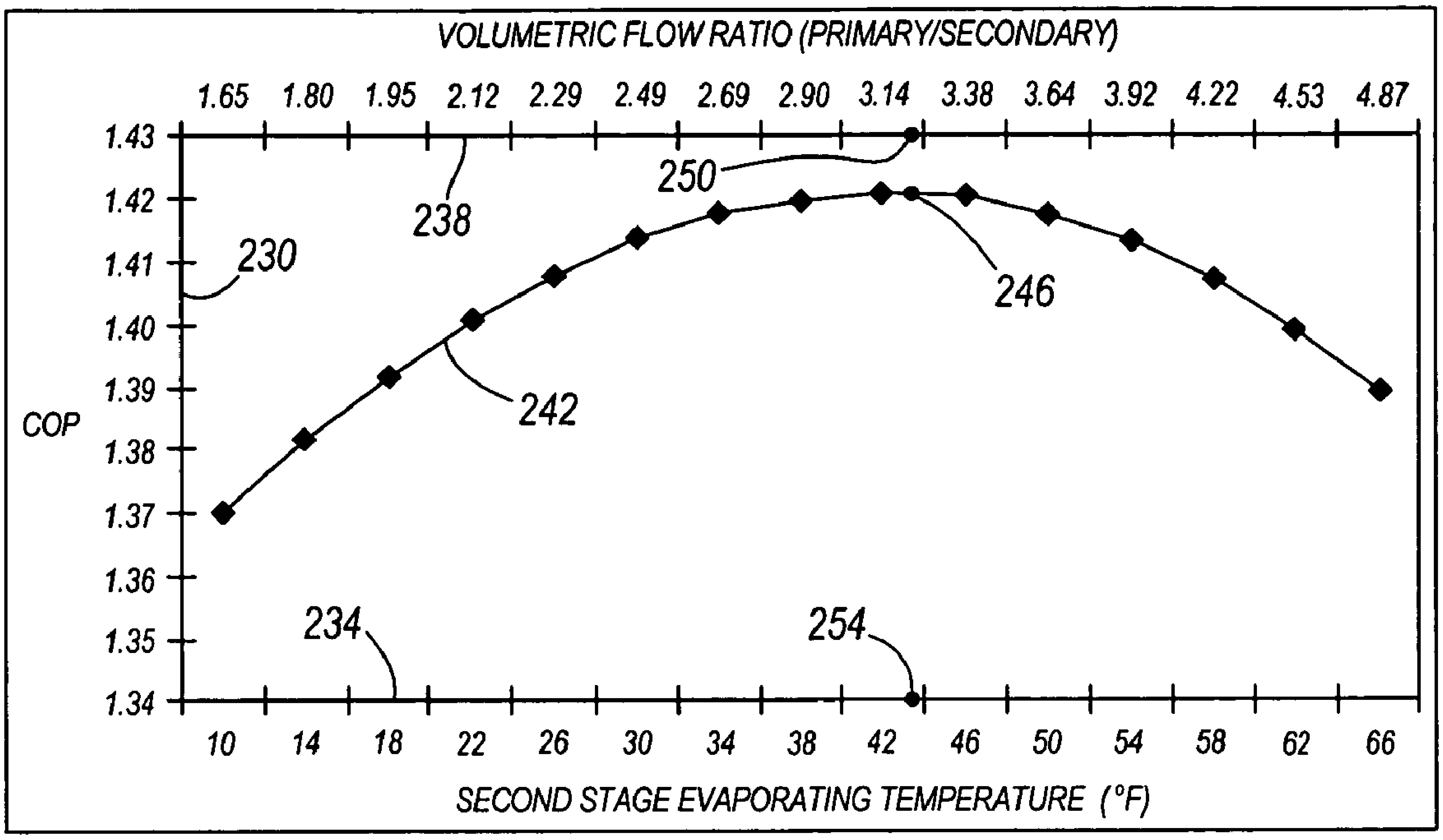
VOLUMETRIC FLOW RATIO (PRIMARY/SECONDARY)
1.65 1.80 1.95 2.12 2.29 2.49 2.69 2.90 3.14 3.38 3.64 3.92 4.22 4.53 4.87
1.43 1.42 1.41 1.40 1.39 1.38 1.37 1.36 1.35 1.34
COP
230
238
250
246
242
234
254
10 14 18 22 26 30 34 38 42 46 50 54 58 62 66
SECOND STAGE EVAPORATING TEMPERATURE (°F)

FIG. 4

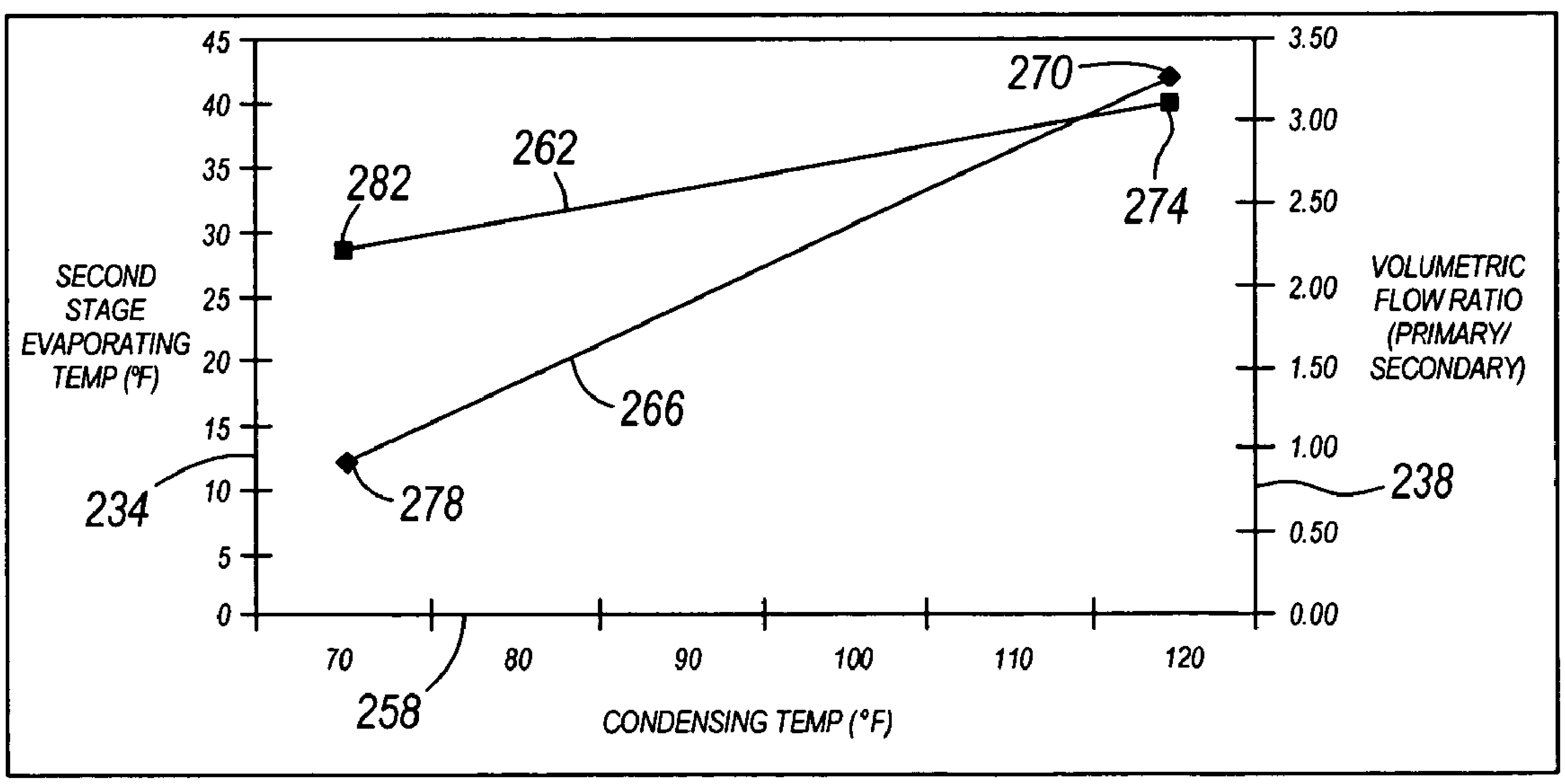
45 40 35 30 25 20 15 10 5 0
SECOND STAGE EVAPORATING TEMP (°F)
3.50 3.00 2.50 2.00 1.50 1.00 0.50 0.00
VOLUMETRIC FLOW RATIO (PRIMARY/ SECONDARY)
270
262
282
274
266
278
234
238
258
70 80 90 100 110 120
CONDENSING TEMP (°F)

FIG. 5

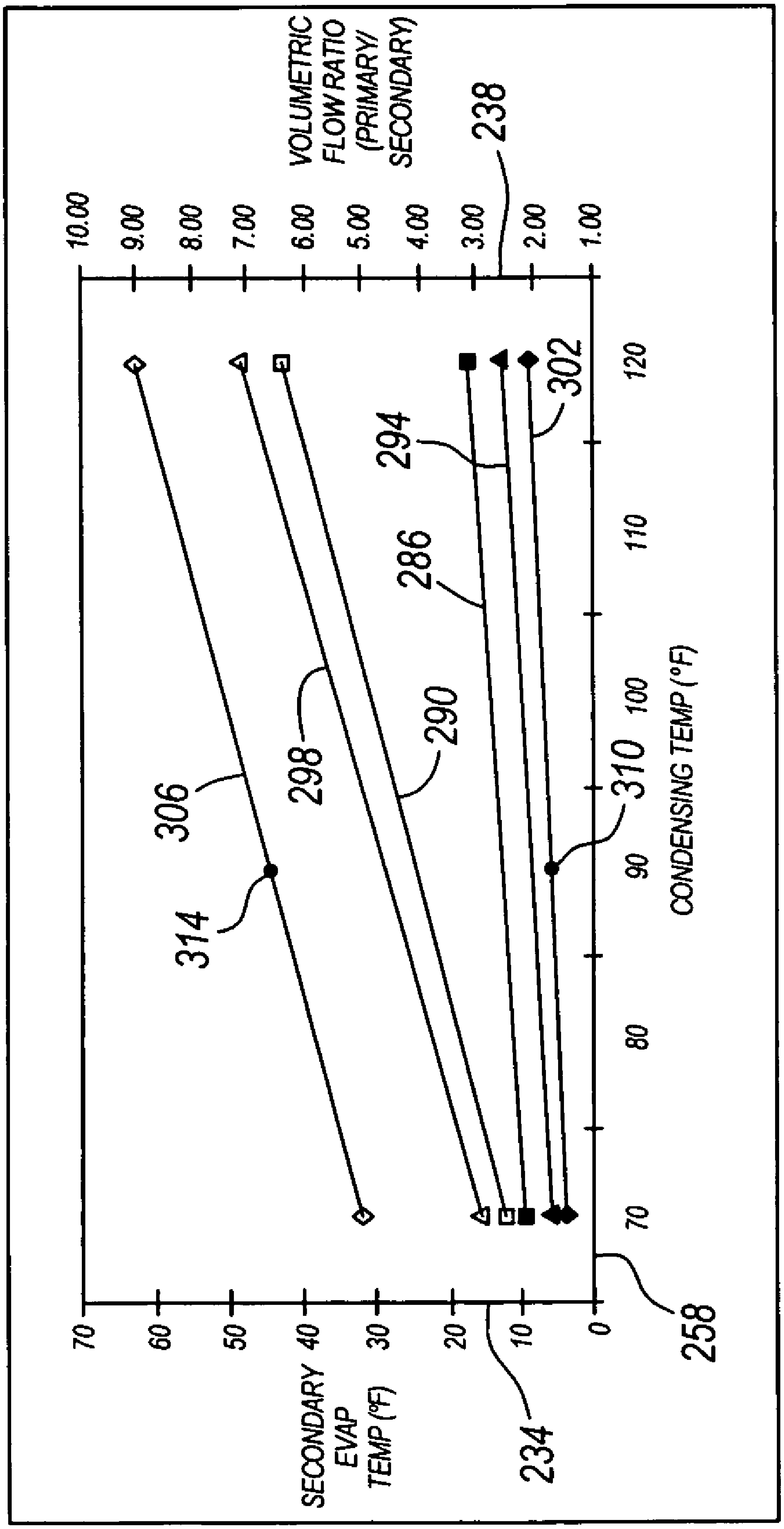
VOLUMETRIC FLOW RATIO (PRIMARY/ SECONDARY)
10.00
9.00
8.00
7.00
6.00
5.00
4.00
3.00
2.00
1.00
238
SECONDARY EVAP TEMP (°F)
70
60
50
40
30
20
10
0
234
258
CONDENSING TEMP (°F)
70
80
90
100
110
120
306
314
298
290
286
294
302
310

FIG. 6

TWO-STAGE LINEAR COMPRESSOR

BACKGROUND

The present invention relates to a refrigeration system including a two-stage linear compressor with dual-opposed pistons, and more particularly to a control system for operating the linear compressor in an economizer cycle.

In refrigeration systems, such as those used in cooling display cases of refrigeration merchandisers, it is necessary to maintain a constant temperature in the display cases to ensure the quality and condition of the stored commodity. Many factors demand varying the cooling loads on evaporators cooling the display cases. Therefore, selective operation of the compressor of the refrigeration system at different cooling capacities corresponds to the cooling demand of the evaporators. In refrigeration systems utilizing existing scroll and screw compressors, an economizer cycle is used to increase the refrigeration capacity and improve efficiency of the refrigeration system. In the economizer cycle of existing scroll and screw compressors, gas pockets in the compressor create a second "piston" as mechanical elements of the compressor proceed through the compression process.

Further, scroll compressors use oil for operation, which results in inefficient performance due to oil film on evaporator and condenser surfaces, requires the use of expensive oil management components, and increases the installation cost of the refrigeration system. Scroll compressors are operable with an economizer, however, efficiency is compromised because the volume ratio is fixed. Some refrigeration systems utilize a linear compressor, which provides variable capacity control of the refrigeration system.

SUMMARY

In one embodiment, the invention provides a control system for managing operation of a dual-piston linear compressor with an economizer cycle wherein a first piston operates as a first stage of the economizer cycle and a second piston operates as a second stage of the economizer cycle. The control system includes a controller coupled to the linear compressor to control a volume flow ratio of the linear compressor. A first sensor for measuring a first operating condition of the linear compressor is coupled to the controller and the first operating condition corresponds to a suction pressure of the linear compressor. A second sensor for measuring a second operating condition of the linear compressor is coupled to the controller and the second operating condition corresponds to a discharge pressure of the linear compressor. A third sensor for measuring a third operating condition of the linear compressor is coupled to the controller and the third operating condition corresponds to an intermediate pressure of the linear compressor. Based upon the first operating condition measured by the first sensor, the second operating condition measured by the second sensor, and the third operating condition measured by the third sensor, the controller varies operation of at least one of the first and second pistons until the intermediate pressure is substantially equal to a pressure required for most efficient operation of the linear compressor.

In another embodiment, the invention provides a control system for managing operation of a dual-piston linear compressor with an economizer cycle wherein a first piston operates as a first stage of the economizer cycle and a second piston operates as a second stage of the economizer cycle. The control system includes a controller coupled to the linear compressor to control a volume flow ratio in the linear compressor, a first sensor for measuring a first operating condition of the linear compressor, and a second sensor for measuring a second operating condition of the linear compressor. The first sensor is coupled to the controller and the first operating condition corresponds to a suction pressure of the linear compressor, and the second pressure sensor is coupled to the controller and the second operating condition corresponds to a discharge pressure of the linear compressor. The controller measures piston stroke of the first piston and piston stroke of the second piston. Based upon the first operating condition measured by the first sensor, the second operating condition measured by the second sensor, and the piston stroke of at least one of the first and second pistons, the controller varies operation of at least one of the first and second pistons until the volume flow ratio is at a point of maximum efficiency.

In yet another embodiment, the invention provides a refrigeration system including a two-stage linear compressor having a first piston disposed in a first cylinder and a second piston disposed in a second cylinder. The linear compressor is operable in an economizer cycle wherein the first piston operates as a first stage of the economizer cycle and the second piston operates as a second stage of the economizer cycle. A controller is coupled to the linear compressor to control a volume flow ratio in the linear compressor. The controller stores a plurality of coefficients of performance for a range of particular operating conditions of the linear compressor, and each coefficient of performance corresponds to a desired volume flow ratio and a desired secondary evaporating temperature. Based upon measured operating conditions of the linear compressor, the controller determines a highest coefficient of performance from the plurality of coefficients of performance and varies operation of at least one of the first and second pistons to achieve the desired volume flow ratio.

Other aspects of the invention will become apparent by consideration of the detailed description and accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a schematic diagram of a refrigeration system including a two-stage linear compressor with dual-opposed pistons embodying the present invention.

FIG. **2** is a schematic diagram of the two-stage linear compressor operating in a single stage cycle.

FIG. **3** is a sectional view of a dual opposing, free-piston linear compressor used in the refrigeration system of FIG. **1**.

FIG. **4** is a chart showing a coefficient of performance (COP) versus secondary evaporating temperature for the refrigeration system and volumetric flow ratio for the linear compressor.

FIG. **5** is a chart showing the volumetric flow rate and secondary evaporating temperature required to maximize COP at a primary evaporating temperature of −40° F.

FIG. **6** is a chart showing the volumetric flow rate and secondary evaporating temperature required to maximize COP at various operating conditions.

Before any embodiments of the invention are explained in detail, it is to be understood that the invention is not limited in its application to the details of construction and the arrangement of components set forth in the following description or illustrated in the following drawings. The invention is capable of other embodiments and of being practiced or of being carried out in various ways. Also, it is to be understood that the phraseology and terminology used herein is for the purpose of description and should not be regarded as limiting. The use of "including," "comprising," or "having" and variations thereof herein is meant to encompass the items listed thereafter and equivalents thereof as well as additional items.

Unless specified or limited otherwise, the terms "mounted," "connected," "supported," and "coupled" and variations thereof are used broadly and encompass both direct and indirect mountings, connections, supports, and couplings. Further, "connected" and "coupled" are not restricted to physical or mechanical connections or couplings.

DETAILED DESCRIPTION

The present invention described with respect to FIGS. **1-6** relates to a control system for operating a two stage linear compressor with dual-opposed pistons in an economizer cycle. The control system controls operation of either or both of a primary piston and a secondary piston of the linear compressor such that a secondary evaporating temperature of the refrigeration system and a volume flow ratio of the linear compressor operate at a point of highest efficiency for the refrigeration system. Generally, the control system varies piston stroke or piston frequency of the primary piston and/or the secondary piston.

FIG. **1** is a schematic diagram of a refrigeration system **10** including a two-stage linear compressor **14** with dual-opposed pistons. In FIG. **1** the linear compressor **14** is shown in an economizer cycle in which refrigerant flows through the refrigeration system along an economizer gas flow path **18** (shown as a bold, solid line in FIG. **1**). In the illustrated embodiment, components of the refrigeration system **10** include the linear compressor **14**, a condenser **22**, an economizer **26** (or liquid subcooler), an expansion device **30** (typically referred to as the expansion valve), and an evaporator **34**, all of which are in fluid communication. In a further embodiment, the refrigeration system **10** includes other components, such as a receiver, a filter, etc.

The refrigeration system **10** includes a controller **38** for controlling operation of the linear compressor **14**. The controller **38** is operable to switch the linear compressor **14** between the economizer cycle (shown in FIG. **1**) and a single stage cycle (shown in FIG. **2**), and to control operation of a primary piston **42** and a secondary piston **46** of the linear compressor **14**. In a further embodiment, one controller operates the linear compressor **14** and another controller operates to switch the linear compressor **14** between the economizer cycle and the single stage cycle.

A schematic of the dual-opposed piston linear compressor **14** is shown in FIGS. **1** and **2**. The linear compressor **14** includes a first cylinder **50** and a second cylinder **54** separated by a dividing wall **58**. The primary piston **42** is disposed in the first cylinder **50** and divides the first cylinder **50** into a suction chamber **62** and a discharge chamber **66**. The primary piston **42** is secured to a spring **70**. Refrigerant enters the suction chamber **62** of the first cylinder **50** from a refrigerant flow path and is discharged from the discharge chamber **66** of the first cylinder **50** to a refrigerant flow path (e.g, the economizer gas flow path **18** shown in FIG. **1** or a single stage gas flow path **74** shown by a bold, solid line in FIG. **2**).

The secondary, or economizer, piston **46** is disposed in the second cylinder **54** and divides the second cylinder **54** into a suction chamber **78** and a discharge chamber **82**. The secondary piston **46** is secured to a spring **86**. The primary and secondary pistons **42**, **46** are opposed and each piston moves back and forth in its respective cylinder in generally opposite directions of movement. Refrigerant enters the suction chamber **78** of the second cylinder **54** from a refrigerant flow path and is discharged from the discharge chamber **82** of the second cylinder **54** to a refrigerant flow path (e.g, the economizer gas flow path **18** shown in FIG. **1** or the single stage gas flow path **74** shown in FIG. **2**). The controller **38** controls piston stroke and displacement or piston frequency (e.g., strokes per second) of the primary and secondary pistons **42**, **46** within the first and second cylinders **50**, **54**. A linear motor (shown in FIG. **3**) for each piston is coupled to the controller **38** and responsive to control signals from the controller **38** to operate the primary and secondary pistons **42**, **46**.

In general, compressed refrigerant discharged from the linear compressor **14** travels to the condenser **22** through a condenser line **90**. After leaving the condenser **22**, the refrigerant next travels to the economizer **26** located upstream of the evaporator **34** through a refrigerant line **94** that divides into a first line **98** and a second line **102**. Refrigerant directed to the first line **98** passes through a first side **106** of the economizer **26** by way of a heat exchanger element (not shown) to the evaporator **34**. After the refrigerant passes through the evaporator **34**, the refrigerant is delivered to the linear compressor **14** through an evaporator line **110**. The controller **38** switches the linear compressor **14** between economizer operation and single stage operation, for example by actuating appropriate control valves positioned in the refrigerant flow paths (e.g, the economizer gas flow path **18** shown in FIG. **1** or a single stage gas flow path **74** shown in FIG. **2**).

When the linear compressor **14** is in the economizer cycle, a portion of the refrigerant is diverted to travel through the second line **102**. The second line **102** is fluidly connected to the expansion valve **30**. Refrigerant directed to the second line **102** passes through the expansion valve **30**, through a second side **114** of the economizer **26**, and out to an economizer line **118**. Refrigerant that passes through the second side **114** of the economizer **26** is used to cool refrigerant that passes through the first side **106** of the economizer **26**. The economizer line **118** delivers refrigerant to the linear compressor **14**. In another embodiment, the refrigerant line **94** divides into a first line and a second line at the outlet of the condenser **22**. In yet another embodiment, the refrigerant line **94** divides into a first line and a second line after the refrigerant exits the first side **106** of the economizer **26**. The first line directs refrigerant to the evaporator **34** and the second line directs refrigerant through the expansion valve **30** and to the second side **114** of the economizer **26**.

In the single stage cycle, refrigerant flows along the single stage gas flow path **74**, shown by the bold line in FIG. **2**. The linear compressor compresses refrigerant in a single step, whereby the refrigerant is compressed by the primary and secondary pistons **42**, **46** with gas flow in parallel. Both the primary piston **42** and the secondary piston **46** share a common suction line **126**, which receives refrigerant from the evaporator line **110**, and a common discharge line **130**, which delivers refrigerant to the condenser line **90**.

In the economizer cycle, refrigerant flows along the economizer gas flow path **18**, shown by the bold, solid line in FIG. **1**. The linear compressor **14** compresses refrigerant in two step process, whereby the refrigerant is compressed first by the primary piston **42** and subsequently by the secondary piston **46**. The suction chamber **62** of the primary piston **42** receives refrigerant from the evaporator line **110**, and the discharge chamber **66** of the primary piston **42** discharges refrigerant to a discharge line **122** that is fluidly connected to the economizer line **118**. The suction chamber **78** of the secondary piston **46** receives refrigerant from the economizer line **118**, which includes refrigerant from both the primary piston chamber **66** and the economizer **26**, and the discharge chamber **82** of the secondary piston **46** discharges refrigerant to the condenser line **90**.

In the economizer cycle, the suction chamber **62** of the primary piston **42** receives cool refrigerant through the evaporator line **110** and the primary piston **42** compresses the refrigerant, which increases the temperature and pressure of the refrigerant. The compressed refrigerant is discharged from the discharge chamber **66** of the primary piston **42** as a warm-temperature, medium-pressure heated gas to the discharge line **122**. Low-temperature, medium-pressure vapor refrigerant from the economizer **26** is mixed with the discharged gas from the primary piston chamber **66** in the economizer line **118**. The mixed refrigerant enters the suction chamber **78** of the secondary piston **46** from the economizer line **118**. Mixing the refrigerant from the primary piston chamber **66** with the refrigerant from the economizer **26** lowers the temperature of the refrigerant entering the secondary piston suction chamber **78**, which prevents overheating of the linear compressor. The secondary piston **46** compresses the mixed refrigerant, which increases the temperature and pressure of the refrigerant. The compressed refrigerant is discharged from the discharge chamber **82** of the secondary piston **46** as a high-temperature, high-pressure heated gas to the condenser line **90**.

The refrigerant travels to the condenser **22** and the condenser **22** changes the refrigerant from a high-temperature gas to a warm-temperature liquid. The high-pressure liquid refrigerant then travels to the economizer **26** through the refrigerant line **94**. A portion of the refrigerant is directed to the first line **98** through the first side **106** of the economizer **22** and the remaining refrigerant is directed to the second line **102** through the second side **114** of the economizer **26**. In one embodiment, a control valve is used to divert refrigerant from the refrigerant line **94** to the second line **102**.

The warm-temperature, high-pressure liquid refrigerant passes through the heat exchanger (i.e., economizer) on the first side **106** and is cooled further to a cool-temperature liquid refrigerant. Warm-temperature, high-pressure liquid refrigerant from the second line **102** passes through the expansion valve **30**, which creates a pressure drop between the second refrigerant line **102** upstream and downstream of the expansion valve **30**. Low-temperature, medium-pressure refrigerant exits the expansion valve **30** and passes through the second side **114** of the economizer **26**, which cools the refrigerant passing through the first side **106** of the economizer **26**.

In the illustrated embodiment, the expansion valve **30** is a thermal expansion valve controlled by pressure and temperature at the outlet of the heat exchanger, i.e., the temperature and pressure in the economizer line **118**. In a further embodiment, the expansion valve **30** is an electronic valve controlled by the controller **38**, or a separate, independent controller (not shown) based upon measured interstage and/or discharge temperature.

The refrigerant from the first side **106** of the economizer **26** enters the evaporator **34** and cools commodities stored in the environmental spaces (not shown). After leaving the evaporator **34**, the cool refrigerant re-enters the suction chamber **62** of the primary piston **42** to be pressurized again and the cycle repeats. The refrigerant from the second side **114** of the economizer **26** enters the economizer line **118** to be mixed with the gas discharged from the discharge chamber **66** of the primary piston **42**. The mixed refrigerant enters the suction chamber **78** of the secondary piston **46** from the economizer line **118** to be pressurized again.

In the economizer cycle, operation of the primary and secondary pistons **42**, **46** is controlled to maintain operation of the linear compressor **14** at a point of best energy efficiency. In particular, the controller **38** controls piston stroke or piston frequency of one or both of the primary and secondary pistons **42**, **46** to maintain a secondary evaporating temperature and a volume flow ratio (i.e., the ratio between the primary piston displacement and the secondary piston displacement) of the linear compressor at values corresponding to a highest efficiency of the refrigeration system **10**. Although the controller **38** controls operation of the linear compressor **14** by either varying piston stroke or varying piston frequency of one or both of the primary and secondary pistons, other known means for controlling operation of the linear compressor to maintain a secondary evaporating temperature and a volume flow ratio may be used.

In one embodiment of the present invention, the controller **38** manages operation of the linear compressor **14** based upon a suction pressure, a discharge pressure, and an intermediate pressure of the linear compressor. As shown in FIG. **1**, the control system includes a first pressure sensor **134**, a second pressure sensor **138**, and a third pressure sensor **142**. The first pressure sensor **134** is disposed in the evaporator line **118** adjacent the linear compressor **14** for measuring a primary suction pressure of the linear compressor **14**. The second pressure sensor **138** is disposed in the condenser line **90** adjacent the linear compressor **14** for measuring discharge pressure of the linear compressor **14**. The third pressure sensor **142** is disposed in the discharge line **122** of the primary piston chamber **66** for measuring intermediate pressure of the linear compressor **14**. All of the sensors **134**, **138**, **142** are coupled to the controller **38** for transmitting the measured pressures to the controller **38**.

In operation, pressure measurements from the first, second, and third pressure sensors **134**, **138**, **142** are transmitted to the controller **38**. The controller **38** stores a plurality of coefficient of performance values (COP) for a range of particular operating conditions of the refrigeration system **10**, in particular an evaporating temperature of the refrigeration system **10** and a condensing temperature of the refrigeration system **10**. The controller **38** derives the evaporating temperature based upon the measured suction pressure and derives the condensing temperature based upon the measured discharge pressure. Based upon the derived evaporating temperature and condensing temperature of the refrigeration system **10**, the controller **38** calculates a COP relating to highest efficiency operation of the linear compressor **14** and the refrigeration system **10** for the specific operating conditions.

The COP cooresponds to a desired secondary evaporating temperature, which corresponds to a desired intermediate pressure, and a desired volume flow ratio for the linear compressor **14**. The controller **38** varies operation of either or both of the primary piston **42** and the secondary piston **46** until the measured intermediate pressure is substantially equal to the desired intermediate pressure needed for highest efficiency of the refrigeration system **10**. For example, if piston stroke of the secondary piston **46** is decreased, the volume flow ratio will increase and the secondary evaporating temperature will increase.

In another embodiment of the control system described above, the first, second and third pressure sensors **134**, **138**, **142** are replaced with sensors that measure other operating conditions of the refrigeration system. For example, a first sensor measures the evaporating temperature of the refrigeration system **10** in the evaporator line **110**, a second sensor measures the condensing temperature of the refrigeration system **10** in the condensing line **90**, and a third sensor measures the secondary evaporating temperature of the refrigeration system **10** in the discharge line **122** from the primary piston chamber **66**.

In another embodiment of the present invention, the controller **38** manages operation of the linear compressor **14** based upon a suction pressure of the linear compressor **14**, a discharge pressure of the linear compressor **14**, and piston stroke of one or both of the primary and secondary pistons **42**, **46**. The control system includes the first pressure sensor **134** disposed in the evaporator line **110** for measuring the suction pressure of the linear compressor **14**, the second pressure sensor **138** disposed in the condenser line **90** for measuring discharge pressure of the linear compressor **14**, and linear motors (shown in FIG. **3**) of the linear compressor **14**. In this embodiment, the third pressure sensor **142** for measuring intermediate pressure is not necessary.

In operation, pressure measurements from the first and second pressure sensors **134**, **138** are transmitted to the controller **38** and the controller **38** measures piston stroke of the primary piston **42** and the secondary piston **46**. As discussed above, the volume flow ratio corresponds to a ratio between piston stroke of the primary piston **42** and piston stroke of the secondary piston **46** (i.e., the ratio between the primary piston displacement and the secondary piston displacement). In one embodiment, the controller **38** infers piston stroke of the primary piston **42** based upon back EMF from the linear motor associated with the primary piston **42**, and the controller **38** infers piston stroke of the secondary piston **46** based upon back EMF from the linear motor associated with the secondary piston **46**.

The controller **38** stores a plurality of COP values for a range of particular operating conditions of the refrigeration system **10**, in particular the evaporating temperature of the refrigeration system **10** and the condensing temperature of the refrigeration system **10**. The controller **38** derives the evaporating temperature based upon the measured suction pressure and derives the condensing temperature based upon the measured discharge pressure. Based upon the derived evaporating and condensing temperatures of the refrigeration system **10**, the controller **38** calculates a COP relating to highest efficiency operation of the linear compressor **14** and the refrigeration system **10** for the specific operating conditions.

In addition to corresponding to a desired secondary evaporating temperature, each COP corresponds to a desired volume flow ratio for the linear compressor **14**. The controller **38** varies operation (e.g., piston stroke or piston frequency) of either or both of the primary piston **42** and the secondary piston **46** until the measured volume flow ratio is substantially equal to the desired volume flow ratio needed for highest efficiency of the linear compressor **14**.

In another embodiment of the control system described above, the first and second pressure sensors **134**, **138** are replaced with sensors that measure other operating conditions of the refrigeration system **10**. For example, a first sensor measures the evaporating temperature of the refrigeration system in the evaporator line **110** and a second sensor measures the condensing temperature of the refrigeration system in the condensing line **90**.

One embodiment of a dual-opposed piston linear compressor **150** is shown in FIG. **3** at an intake stroke. The dual-opposed piston linear compressor **150** includes a housing **154** supporting a main body block **158**. Inner and outer laminations **162** and **166** are secured to the main body block **158** and coils **170** are wound on the outer laminations **166**, thereby resulting in stators. The stators, when energized, interact with magnet rings **174** mounted on outer cylinders **178**. The outer cylinders **178** are fastened to a first piston **182** and a second piston **186**, which are secured to springs **190**. The interaction between the magnet rings **174** and the energized stators results in the outer cylinders **178** moving the pistons **182**, **186** linearly along an axis of reciprocation **194**. A linear motor for each piston is defined by the stator and the magnet rings **174**.

A dividing wall **198** separates the first piston **182** and the second piston **186** into a first chamber **202** and a second chamber **206**, respectively. Each chamber includes a suction portion **202***a* and **206***a* and a compression portion **202***b* and **206***b*, or discharge portion. When the first and second pistons **182**, **186** are at the intake stroke, refrigerant is allowed to flow from a suction port **210** at the suction portion **202***a*, **206***a* of each chamber **202**, **206** through channels **214** to the compression chambers **202***b*, **206***b*. When moving from the intake stroke to a compression stroke, the channels **214** are closed by suction valves **218** and refrigerant is compressed out of the compression chambers **202***b*, **206***b* through discharge valves **222** and discharge ports **226**.

The linear motor allows for variable compression by the pistons **182**, **186**, and therefore, the linear compressor **150** provides variable capacity control. In other words, the linear motors can cause the pistons **182**, **186** to move a small stroke for a first volume, or to move a larger stroke for a second, larger volume.

In a further embodiment of the linear compressor **14**, the primary piston **42** has a larger displacement than the secondary piston **46** to increase the compression ratio of the linear compressor **14** and increase the density of the refrigerant discharged from the linear compressor **14**. For example, the primary piston **42** has a larger diameter than the secondary piston **46** or the primary piston **42** has a longer piston stroke than the secondary piston **46**. In one embodiment, piston stroke of the primary and secondary pistons **42**, **46** is adjusted by the controller **38**, and in another embodiment piston frequency of the primary and secondary pistons **42**, **46** are adjusted by the controller **38**.

FIGS. **4**-**6** are charts illustrating an example of the methodology used by the controller **38** to determine maximum efficient operation of the linear compressor **14** and the refrigeration system **10**. The charts illustrated in FIGS. **4**-**6** reflect the use of R410A refrigerant in the refrigeration system **10**, which is a chlorine-free refrigerant. It should be readily apparent that other types of refrigerant may be used in the refrigeration system **10**.

FIG. **4** is a chart showing a coefficient of performance (COP) **230** versus secondary evaporating temperature **234** for the refrigeration system **10** and volume flow ratio **238** for the linear compressor **14**. FIG. **4** is directed to a specific operating condition of the refrigeration system, −40° F. evaporating temperature and 120° F. condensing temperature. COP **230** relative to the operating condition of the refrigeration system is shown on the Y-axis, and the X-axis has two scales, 1) the secondary evaporating temperature **234** corresponding to a particular COP, and 2) the volume flow ratio **238**. As shown in FIG. **4**, line **242** represents the COPs for the specific operating condition of the refrigeration system **10** and the COP is highest at point **246** when the volume flow ratio is 3.2 (point **250**), which corresponds to a 44° F. secondary evaporating temperature (point **254**).

FIG. **4** illustrates that operation of the refrigeration system **10** can be optimized by controlling the secondary evaporating temperature and the volume flow ratio between the primary and secondary pistons **42**, **46**. As discussed above with respect to the control systems, the refrigeration system **10** controls the secondary evaporating temperature and the volume flow ratio by varying operation of either or both of the primary and secondary pistons **42**, **46**.

FIG. **5** is a chart showing the volumetric flow rate and secondary evaporating temperature required to maximize COP at a primary evaporating temperature of −40° F. and thereby operate the refrigeration system **10** at highest efficiency. Condensing temperature **258** for the refrigeration system **10** is shown on the X-axis, and the secondary evaporating temperature **234** and the volume flow ratio **238** are shown on the two Y-axes. Line **262** corresponds to the volume flow ratio at −40° F. evaporating temperature and various condensing temperatures, and line **266** corresponds to the secondary evaporating temperature at −40° F. evaporating temperature and various condensing temperatures. Lines **262** and **266** indicate the volume flow ratio and the secondary evaporating temperature needed for highest efficiency. For example, at −40° F. evaporating temperature and 120° F. condensing temperature, the desired secondary evaporating temperature is 44° F. (point **270**) and the desired volume flow ratio is 3.2 (point **274**) to obtain the highest efficiency (also shown by FIG. **4**). As another example, at −40° F. evaporating temperature and 70° F. condensing temperature, the desired secondary evaporating temperature is 12° F. (point **278**) and the desired volume flow ratio is 2.2 (point **282**) to obtain the highest efficiency. For any condensing temperature at −40° F. evaporating temperature, the highest efficiency secondary evaporating temperature and volume flow ratio can be found by selecting the appropriate points on the graph.

FIG. **6** is a chart showing the volumetric flow rate and secondary evaporating temperature required to maximize COP at other evaporating conditions. The condensing temperature **258** for the refrigeration system is shown on the X-axis, and the secondary evaporating temperature **234** and the volume flow ratio **238** are shown on the two Y-axes. In FIG. **6**, line **286** corresponds to the volume flow ratio at −40° F. evaporating temperature, and line **290** corresponds to the secondary evaporating temperature at −40° F. evaporating temperature. Line **294** corresponds to the volume flow ratio at −25° F. evaporating temperature, and line **298** corresponds to the secondary evaporating temperature at −25° F. evaporating temperature. Line **302** corresponds to the volume flow ratio at 0° F. evaporating temperature, and line **306** corresponds to the secondary evaporating temperature at 0° F. evaporating temperature. Accordingly, the most efficient secondary evaporating temperature and volume flow ratio can be found for many operating conditions by locating the appropriate point in FIG. **6**. For example, at 0° F. evaporating temperature and 90° F. condensing temperature, the desired volume ratio is 1.6 (point **310**) and the desired secondary evaporating temperature is 41° F. (point **314**) for highest efficiency operation of the refrigeration system **10**.

The controller **38** determines maximum efficient operation of the linear compressor **14** and the refrigeration system **10** using the factors and methodology described above with respect to FIGS. **4**-**6**. The controller **38** stores a plurality of COPs for a variety of operating conditions for the refrigeration system **10**. Based upon the factors measured and received by the controller **38**, such as suction pressure, discharge pressure, and intermediate pressure (or temperature) or piston stroke of the primary and secondary pistons **42**, **46**, the controller **38** references a highest COP for the corresponding evaporating temperature and condensing temperature. The COP corresponds to a secondary evaporating temperature and a volume flow ratio for highest efficiency operation of the refrigeration system **10**. The controller **38** adjusts piston stroke or piston frequency of either or both of the primary and secondary piston **42**, **46** to achieve the desired secondary evaporating temperature and desired volume flow ratio.

Various features and advantages of the invention are set forth in the following claims.

What is claimed is:

1. A control system for managing operation of a dual-piston linear compressor with an economizer cycle wherein a first piston operates as a first stage of the economizer cycle and a second piston operates as a second stage of the economizer cycle, the control system comprising:

a controller coupled to the linear compressor to control a volume flow ratio of the linear compressor and to vary the linear compressor between the economizer cycle having a first gas flow path through the linear compressor and a single stage cycle having a second gas flow path through the linear compressor, the first gas flow path partially defined by the first piston receiving gas from an evaporator line and discharging gas to an economizer line, and further partially defined by the second piston receiving gas from the economizer line and discharging gas to a condenser line such that a suction chamber of the second stage is in fluid communication with a discharge chamber of the first stage;

a first sensor coupled to the controller to measure a first operating condition of the linear compressor including a suction pressure of the linear compressor;

a second sensor coupled to the controller to measure a second operating condition of the linear compressor including a discharge pressure of the linear compressor; and

a third sensor coupled to the controller to measure a third operating condition of the linear compressor including an intermediate pressure of the linear compressor,

wherein based upon the first operating condition measured by the first sensor, the second operating condition measured by the second sensor, and the third operating condition measured by the third sensor, the controller varies operation of at least one of the first and second pistons until the intermediate pressure is substantially equal to a pressure required for most efficient operation of the linear compressor.

2. The control system of claim **1** wherein the first sensor measures the suction pressure of the linear compressor.

3. The control system of claim **1** wherein the second sensor measures the discharge pressure of the linear compressor.

4. The control system of claim **1** wherein the third sensor measures the intermediate pressure of the linear compressor.

5. The control system of claim **1**, wherein the controller calculates a secondary evaporating temperature required for most efficient operation of the linear compressor based upon the suction pressure and the discharge pressure.

6. The control system of claim **5**, wherein the controller stores a plurality of coefficients of performance for a range of particular operating conditions of the linear compressor, each coefficient of performance corresponding to a desired intermediate pressure, and further wherein the controller determines a highest coefficient of performance from the plurality of coefficients of performance and varies operation of at least one of the first and second pistons to achieve the desired secondary evaporating temperature.

7. The control system of claim **1**, wherein the controller varies operation by adjusting piston stroke for at least one of the first and second pistons.

8. The control system of claim **1**, wherein the controller varies operation by adjusting piston frequency for at lest one of the first and second pistons.

9. A control system for managing operation of a dual-piston linear compressor with an economizer cycle wherein a first piston operates as a first stage of the economizer cycle and a second piston operates as a second stage of the economizer cycle, the control system comprising:

a controller coupled to the linear compressor to control a volume flow ratio of the linear compressor and to vary the linear compressor between the economizer cycle having a first gas flow path through the linear compressor and a single stage cycle having a second gas flow path through the linear compressor, the first gas flow path partially defined by the first piston receiving gas from an evaporator line and discharging gas to an economizer line, and further partially defined by the second piston receiving gas from the economizer line and discharging gas to a condenser line such that a suction chamber of the second stage is in fluid communication with a discharge chamber of the first stage;

a first sensor coupled to the controller to measure a first operating condition including a suction pressure of the linear compressor;

a second sensor coupled to the controller to measure a second operating condition including a discharge pressure of the linear compressor; and

wherein the controller measures piston stroke of the first piston and piston stroke of the second piston, and

further wherein based upon the first operating condition measured by the first sensor, the second operating condition measured by the second sensor, and the piston stroke of at least one of the first and second pistons, the controller varies operation of at least one of the first and second pistons until the volume flow ratio is at a point of maximum efficiency.

10. The control system of claim **9** wherein the first sensor measures the suction pressure.

11. The control system of claim **9** wherein the second sensor measures the discharge pressure.

12. The control system of claim **9** wherein the linear compressor includes a first linear motor for causing displacement of the first piston and a second linear motor for causing displacement of the second piston, and further wherein the controller infers the piston stroke of at least one of the first and second pistons based upon back EMF from the linear motor associated with the piston.

13. The control system of claim **9** wherein the controller calculates the volume flow ratio required for maximum efficiency based upon the suction pressure, the discharge pressure and the piston stroke of at least one of the first and second pistons.

14. The control system of claim **9**, wherein the controller stores a plurality of coefficients of performance for a range of particular operating conditions of the linear compressor, each coefficient of performance corresponding to a desired volume flow ratio, and further wherein the controller determines a highest coefficient of performance from the plurality of coefficients of performance and varies operation of at least one of the first and second pistons to achieve the desired volume flow ratio.

15. The control system of claim **9**, wherein the controller varies operation by adjusting piston stroke of at least one of the first and second pistons.

16. The control system of claim **9**, wherein the controller varies operation by adjusting piston frequency of at least one of the first and second pistons.

17. A refrigeration system comprising:

a two-stage linear compressor including a first piston disposed in a first cylinder and a second piston disposed in a second cylinder, the linear compressor operable in an economizer cycle wherein the first piston operates as a first stage of the economizer cycle and the second piston operates as a second stage of the economizer cycle;

a controller coupled to the linear compressor to control a volume flow ratio in the linear compressor and to vary the linear compressor between the economizer cycle having a first gas flow path through the linear compressor and a single stage cycle having a second gas flow path through the linear compressor, the first gas flow path partially defined by the first piston receiving gas from an evaporator line and discharging gas to an economizer line, and further partially defined by the second piston receiving gas from the economizer line and discharging gas to a condenser line such that a suction chamber of the second stage is in fluid communication with a discharge chamber of the first stage,

wherein the controller stores a plurality of coefficients of performance for a range of particular operating conditions of the linear compressor, each coefficient of performance corresponding to a desired volume flow ratio and a desired secondary evaporating temperature, and

further wherein based upon measured operating conditions of the linear compressor the controller determines a highest coefficient of performance from the plurality of coefficients of performance and varies operation of at least one of the first and second pistons to achieve either the desired volume flow ratio or the desired secondary evaporating temperature.

18. The refrigeration system of claim **17** wherein the controller varies operation of at least one of the first and second pistons based upon a suction pressure and a discharge pressure.

19. The refrigeration system of claim **18** wherein the controller varies operation of at least one of the first and second pistons until a measured intermediate pressure is substantially equal to an intermediate pressure corresponding to the desired secondary evaporating temperature.

20. The refrigeration system of claim **17** wherein the controller varies operation of at least one of the first and second pistons based upon a suction pressure, a discharge pressure, and a measured piston stroke of at least one of the first and second pistons.

21. The refrigeration system of claim **20** wherein the linear compressor includes a first linear motor for causing displacement of the first piston and a second linear motor for causing displacement of the second piston, and further wherein the measured piston stroke is inferred from back EMF of the linear motor of the at least one piston.

22. The refrigeration system of claim **17**, and further comprising:

a first pressure sensor for measuring suction pressure of the linear compressor; and

a second pressure sensor for measuring discharge pressure of the linear compressor,

wherein the first pressure sensor and the second pressure sensor are electrically connected to the controller.

23. The refrigeration system of claim **22** wherein the controller is operable to measure piston stroke of the first piston and piston stroke of the second piston.

24. The refrigeration system of claim **23** wherein the controller calculates a volume flow ratio at maximum efficiency based upon suction pressure measured by the first pressure sensor, discharge pressure measured by the second pressure sensor, and the measured piston stroke of at least one of the first and second pistons.

25. The refrigeration system of claim **24** wherein the controller varies operation of at least one of the first and second pistons to achieve the volume flow ratio at maximum efficiency.

26. The refrigeration system of claim **22**, and further comprising a third pressure sensor for measuring intermediate pressure of the linear compressor, wherein the third pressure sensor is electrically connected to the controller.

27. The refrigeration system of claim **26** wherein the controller is operable to vary operation of at least one of the first and second pistons based upon suction pressure measured by the first pressure sensor and discharge pressure measured by the second pressure sensor until the measured intermediate pressure is substantially equal to an intermediate pressure needed for maximum efficiency.

28. The refrigeration system of claim **17**, wherein the controller varies operation by adjusting piston stroke of at least one of the first and second pistons.

29. The refrigeration system of claim **17**, wherein the controller varies operation by adjusting piston frequency of at least one of the first and second pistons.

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